JOHNSON

Centrifugal and Rotary Pumps

Mech. Engineering
B. S.

1903

# CENTRIFUGAL AND ROTARY PUMPS

BY

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THESIS FOR THE DEGREE OF BACHELOR OF SCIENCE
IN MECHANICAL ENGINEERING

IN THE
COLLEGE OF ENGINEERING
OF THE
UNIVERSITY OF ILLINOIS
PRESENTED JUNE, 1903

UNIVERSITY OF ILLINOIS

June 1, 1903 190

THIS IS TO CERTIFY THAT THE THESIS PREPARED UNDER MY SUPERVISION BY

ALBERT MYRON JOHNSON

ENTITLED ROTARY AND CENTRIFUGAL PUMPS

IS APPROVED BY ME AS FULFILLING THIS PART OF THE REQUIREMENTS FOR THE DEGREE

OF Bachelor of Science in Mechanical Engineering.

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# CENTRIFUGAL AND ROTARY PUMPS

## I. INTRODUCTORY REMARKS

Although some investigation has been carried on along the lines of Centrifugal and Rotary pumps, still the subject furnishes a wide field for study.

It is the object of this thesis to compare the two types of pumps, to show the uses to which each is best adapted, and to give their efficiencies and the power absorbed by them under similar circumstances.

First, there will be given an account of the early rotary pumps and their subsequent development; second, a similar discussion of the centrifugal type will be given.

## II. HISTORY

### a. Rotary Pumps

The rotary pump is a machine which drives the liquid by the direct action of the rotating impellers or pistons. These impellers fit the casing in which they turn, and for this reason

it
forms a positive action pump. That is, if there were no leakage or slip the amount of fluid pumped would vary directly as the speed of the pump.

Probably the first rotary pump was invented by Ramelli in 1588. It consisted of a hub (d) (Fig. 1) placed eccentrically within a cylinder (a), so that their peripheries nearly touched at a point midway between the suction and delivery pipes. Four rectangular blocks or pistons (c) were arranged to travel in radial slots cut across the hub. The pistons were kept against the cylinder shell by means of coiled springs. The water entering by the suction pipe was carried around, and as it could not pass by the abutment was forced out of the discharge.

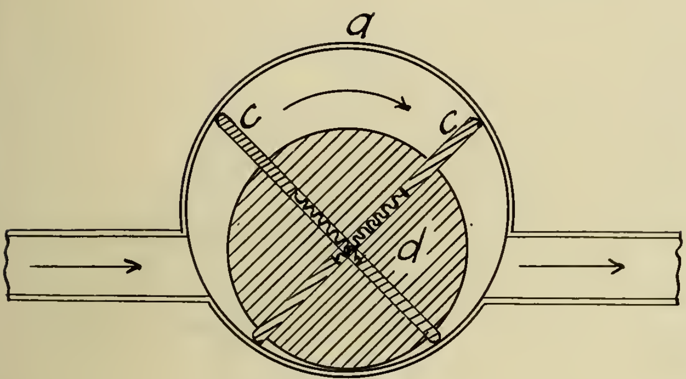



Fig. 1.

Another and perhaps more famous pump, known as the "Pappenheim Rotary Pump" (Fig. 2) was invented in 1636. This pump is composed of two gears, which mesh within an elliptical casing.

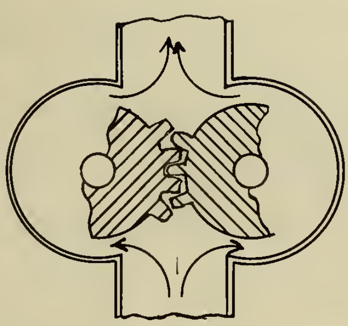


Fig 2.

The sides of the gears run closely against the sides of the casing, and their peripheries have just enough clearance to turn freely within the semi-circular ends of the casing. This type, although one of the oldest is one of the best ever invented, and it is seen in improved form in many pumps in use at the present time.

The Root blower (Fig. 3), which in principle is exactly like the rotary pump, is a development of the above type. It differs only in the number of teeth on the gear wheels, in this case there being but two lobes or teeth to each gear. The shafts to which these impellers are fastened must, of course, be paired together by outside gears. The surfaces of the impellers are epi-, and hypo-cycloids so there is a rolling motion at their point of contact.

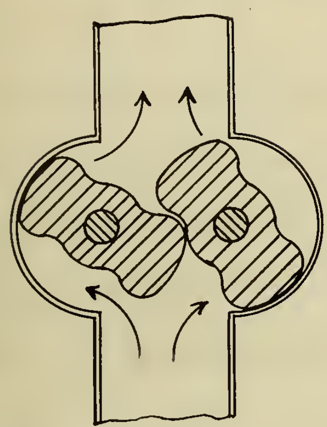


Fig 3.

The last step in reducing the number of teeth of the Pappenheim pump, is represented in the Repsold pump(Fig.4). In this form, there is much sliding between the point and root cylinders, and consequently there is much wear.

The next pump deserving mention is the "Emery" (Fig.5). A cylindrical driving hub placed eccentrically within the cylindrical casing and having four impellers sliding in radial slots, are the main features of this pump. The

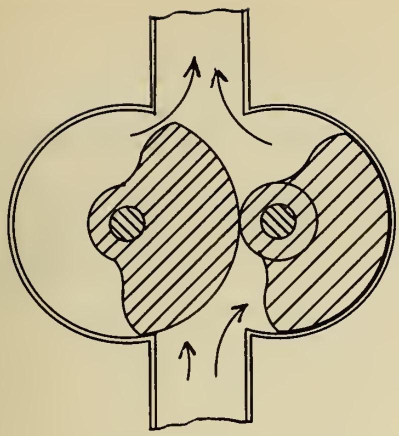


Fig. 4.

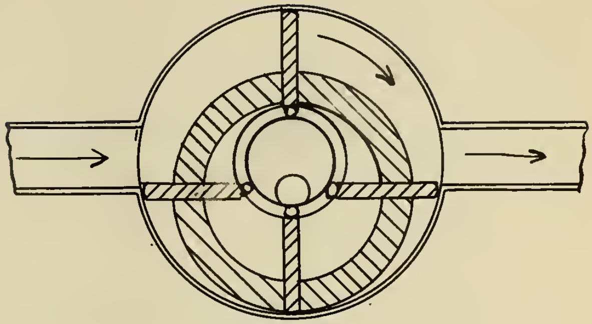


Fig 5.

rectangular pistons are very similar to those used in the Ramelli pump but are held against the casing by grooves concentric with the casing. This principle was applied to a pump of recent date, but was a failure because of the large number of wearing parts and the heavy pressures brought upon the grooves with their small bearing surfaces.

Another interesting type of pump is that known as the Pattison, which was patented in England in 1857. The impeller has the form of an eccentric; the strap fastened to an arm engages a sliding block by a socket joint. The arm and sliding block, as will be seen by the sketch (Fig 6) form the separating element between the suction and discharge. This pump was followed soon after by the Knott, and the Bartram & Powell pumps, both following the same general principle as the one just described.

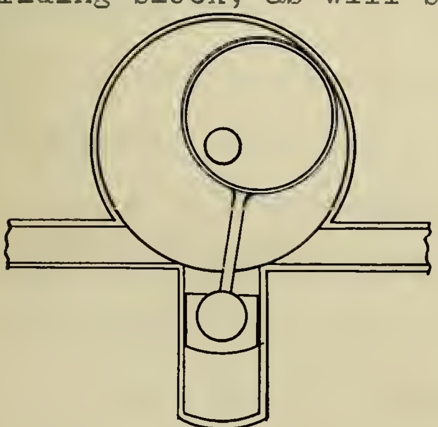


Fig 6.

There have been a great many other rotary pumps patented, but most of them have been of such a complicated or impracticable nature, that they have never come into extensive use.

### b. The Centrifugal Pump

The first centrifugal pump was probably brought out by the mathematician Euler, in 1753. The next pump approaching the efficiencies now obtained was built by McCarty at the New York Navy Yard in 1830.

In 1851 followed the celebrated Appold pump, the first pump upon which authentic tests were made. Mr. Appold determined that the efficiency mainly depended upon the form of blades in the fan and the shape of the enveloping case, and that the best form of blades was a curve pointing in direction opposite to that in which the fan revolved, and for the caseing, that of a spiral tapering pipe or volute.

These determinations were arrived at from the following considerations:- 1.- The water must have as high a velocity as possible as it leaves the blades. 2.- This velocity must be reduced, thus changing the velocity head into pressure head.

In 1875 some experiments were made in England. These tests were run, to show what effect alterations of the runner and casing would have upon the performance of the pump.

The first experiments were run with an Appold runner in a circular casing. The Appold runner was composed of vanes curving constantly in a direction opposite to the direction of

rotation.

The Appold runner is represented at A (Fig. 7). The first tests were made with the casing in circular form, then with the casing in spiral form, represented by the dotted line. This spiral form was made by fitting in a block of wood C. It was found by comparing the two sets of tests that the set made with the spiral casing was greatly improved in efficiency, and also in increase of discharge.

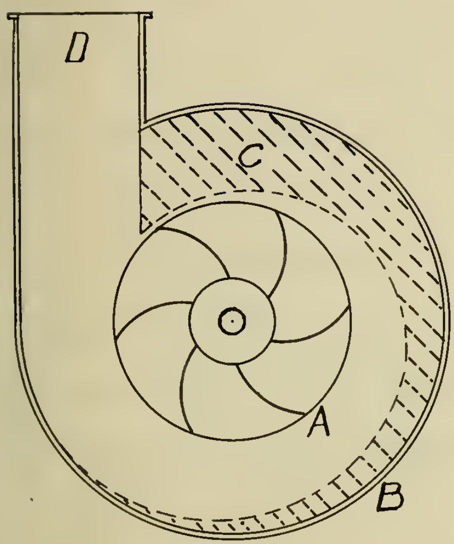


Fig. 7.

Rankine worked out a form of blade, which he considered mathematically correct, to give the maximum efficiency. These two forms of blades have caused much discussion; and there is a wide difference of opinion as regards which one is the better. The Rankine blade, (Fig.8) has a compound curve, the curve at the periphery being radial[1]. This he reasoned would give the water a maximum velocity as it left the blade, a condition which was essential for an efficient pump. Experiment

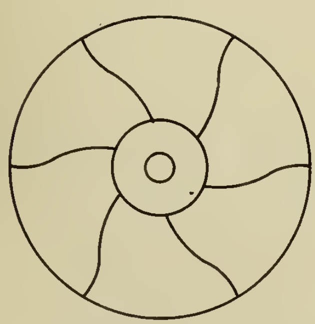


Fig. 8.

1 Trans. A. S. M. E. Vol. 9, page 228. William O Webber.

has borne him out to this extent. The pump will give a very high efficiency, if run at the speed and under the head, for which it was designed. These conditions are seldom met in practice, and, this being the case, a runner had to be used which would give a high efficiency even if run under varying conditions. Thus the Appold in general form is the one most extensively used.

The development of the centrifugal pump, since Appold brought out his pump, has been, for the most part, in the better design of the runner. The efficiency of specially designed pumps has gradually risen from 25% to 75%.

# III DESCRIPTION OF MODERN PUMPS

## a. Rotary Pumps

The rotary pump, as distinguished from the centrifugal pump, is positive in action; i. e. the discharge varies directly as the speed, neglecting slip.

The fluid is driven by the direct pressure of the propeller pistons, of some sort, which closely fit the casing; thus preventing the discharge from passing back into the suction in any way other than by the leakage between the piston and casing.

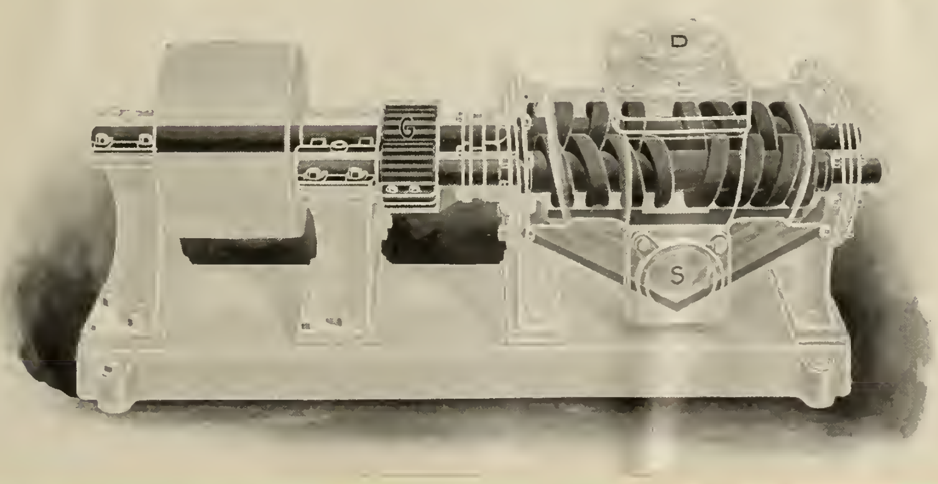
The modern rotary pumps, for the most part, are developments of the pumps described in the early part of this thesis. One of the very common types, especially in the small sizes is that class which is patterned after the Pappenheim pump; being simply two spur gears meshing together inside of a casing. This pump produces pulsations in its discharge and is not manufactured in sizes much larger than that having two inch suction and discharge.

The Root Pump is probably the one used to the greatest extent at present. This pump (Fig. 9) as previously described is a development of the Pappenheim principle, in so far as the number of teeth on each spur gear is reduced from <u>n</u>

in the Pappenheim to two in the Root. The surfaces roll upon each other thus reducing wear to a minimum. The impellers are driven by external gears

Fig. 9.

The Quimby Screw Pump is another development of the Pappenheim. The teeth of the gears instead of being parallel are wound around the axis, forming a square threaded screw. Its action can readily be understood from the following engraving



The claims made for this pump are (a) its capability in pumping against high heads; (b) the absence of thrust on the shaft journals; and (c) its high efficiency. The objections to it are: its liability to wear, and its very high first cost.

The Johnson Rotary Pump is a somewhat wider variation of the Pappenheim pump. In this pump one large drum carries four teeth on its periphery, and a second drum one-fourth as large has on its circumferance one cavity for the teeth of the large drum. This pump will be described more fully later in connection with the report of tests made upon it.

The design, in principle, of a rotary pump involves simply the design of a machine which will cause the fluid to pass from the suction to the discharge of the pump and either by rotating or stationary abutments, prevent it from flowing back into the suction.

There are many mechanical difficulties to be considered, the chief one of which is the wear between the various moving parts, and as a result the pump gives a low efficiency and is very short lived.

The well designed rotary pump should show an efficiency exceeding the best centrifugal pump. This is due to the fact that the latter, as will be shown later, has many friction losses between the rotating parts and the water; and between the water and the casing.

The rotary pump is used where a positive action pump is necessary, and under other conditions similar to those under which the centrifugal pump operates. One of the chief advantages of the rotary pump lies in the fact that it will discharge at any speed whatever, within mechanical limits, of course. The centrifugal pump, on the other hand, will not discharge until the speed is brought up to a certain amount, depending upon the lift and the shape of the casing and vanes.

In general, the first cost of the rotary pump is much higher than that of the centrifugal pump. This consideration often causes the prospective purchaser to decide in favor of the latter type.

## b. Centrifugal Pumps

All of the centrifugal pumps manufactured at the present time may be placed in two classes:- First. - Those pumps, usually of large size, especially designed for certain conditions of service in connection with some important work. Second. - Pumps known as "commercial article" not built for particular conditions, but being rather a compromise in design to meet varying conditions.

Many of the latter class are used for dredging purposes and for handling various solids in suspension.

Centrifugal pumps are usually designated as to size by diameter of discharge. Their discharge is given in gallons per minute at economical capacity.

The use of this class of pump in city water works service is gradually displacing that of the reciprocating type for small installations. It is well known that the most economical pumping engine is not the one showing the highest efficiency in terms of coal consumption, but that in which the sum of the fixed charges and coal cost is least. These considerations would lead us to the conclusion that a well designed system using centrifugal pumps having a capacity up to 2,500,000 gallons could be built and operated more economically than any other system. Especially is this the case where cheap electrical power can be supplied. It can pump directly into the mains always furnishing any amount of water up to its capacity, and at but slightly varying pressure. If the discharge is brought to zero by a closing of all the service cocks, the pump continues to run with no effect, with the exception of a slight increase in pressure; and when the discharge is large the pressure falls a small amount. Thus it will be seen that it is self regulating, and if electrically driven by an induction motor, does not require constant attendance, for the unit as a whole will adapt itself to conditions. By arranging two or more pumps on the same shaft, and by the use of by-passes, the pressure can be readily increased for fire purposes by allowing the discharge of one pump to flow into the suction of the second through the by-pass. A pulley may be placed on the pump shaft to be driven by some other source of power in case the electric motor

should fail. The irrigation of the rice fields in the South furnishes another important use to which this form of pump may be well applied. The conditions under which it operates are: Low head and large discharge. This same class of pump, on a larger scale, is much used for drainage and sewerage purposes. Especially has this type been developed in Italy and The Netherlands, for use in the reclaiming of low marshy land.

Within the last few years pumps have been designed to work under heads from two hundred to one thousand feet, by compounding, which method will be described later. Some efficiencies approaching the maximum have been obtained with this class. They are used chiefly for mining purposes. From the foregoing it will be seen that, in general, the centrifugal pump may be used advantageously where large quantities of liquid are to be moved against any reasonable head.

The work to be performed by a centrifugal pump is equal to the work of lifting water from the pit to the delivery level plus the work of giving kinetic energy or energy of motion to the flowing water. The first is the net useful effect dependent upon the speed and the shape of the vanes, and cannot be attained until the peripheral speed of the runner is brought up to a given amount. The second item of the work is that work necessary to overcome the frictional resistance to the flow through the pipes and passages.

The frictional resistance of an ordinary centrifugal pump with enclosed runner and running without discharge, may

be divided into four parts:-[1]

a. The slip, or back circulation around the suction inlet caused by leakage of water from the discharge under a head due to the static lift. As ordinarily manufactured with leakage on both sides of the runner and an opening of one thirty-second of an inch around the suction inlet and balance ring, this one item may account for twenty per cent of lost work.

b. The frictional resistance of the water circulated across the side plates and through the vanes of the runner, varies approximately as the square of the velocity of flow and as the flow is proportional to the square root of h, this item also decreases with the lift.

c. The drag of the runner periphery revolving at a high speed in water, practically without velocity. Assume speed of runner periphery to be eighty feet per second. When discharge is taking place, the water in the throat and volute of the pump will attain a velocity proportional to the area. Let us assume a velocity of flow of fifteen feet per second in the same direction as that of the runner. The loss then would be decreased approximately in the ratio of sixty-five squared, to eighty squared. The difference representing about forty per cent of the original resistance. As this loss decreases with the increase in velocity of flow, it evidently decreases with the lift.

d. The friction of the runner with side plates. Since

[1]- From paper read before Pacific Coast Electric Transmission Association. Eng. News, August 9, 1900.

this factor is independent of the pressure it will be a constant quantity for any height of lift. When discharge begins conditions are somewhat changed since new elements of resistance are introduced. These are made up of the head necessary to create velocity of flow past the resistance offered by the pipes and pump passages together with the kinetic energy lost in the form of discharge velocity. The sum of these ratios varies probably in the ratio somewhat less than the square of the velocity of flow and therefore increases with an increase in lift.

A careful consideration of these factors of resistance would lead to the conclusion that, as the lift decreases the decrease in friction would be in excess of the increase due to velocity. Hence it follows that a pump discharging against a low head would have a higher efficiency than one which must pump against a high head. However, experiment seems to show that the most efficient pumps are those which work against a high pressure[1].

According to Mr. Edward S. Cobb of San Francisco the discharge of a centrifugal pump at constant speed, for any lift, varies as the square root of the difference between the actual lift and the hydrostatic head created by the pump without discharge.

The peripheral runner speed in terms of the height of lift is expressed by the following formula:-

1.- See page 27.

$\pi d n = 60 \times 10\sqrt{h}$.

According to one of the best hand books on hydraulics, the tangential runner velocity if feet per minute required to hold water at any height without discharge is equal to five hundred times the square root of $h$. This value may perhaps be taken as an average for a large number of pumps, but it has been found by experiment that this factor varies from about three to eight hundred times the square root of $h$, according to the shape of the vanes. It follows then that for a constant speed there must be a form of vane more suitable than any other.

The discussion used in the consideration of the action of the centrifugal pump is, in a manner, similar to that used in tracing the effect of the driving water on the floats of turbines. Let AB (Fig. 11) be a blade which makes with the inner circle of radius $r_1$, and outer circle of radius $r_2$ the angles $\alpha$ and B respectively. Assuming that water is led to the inner circle in radial direction, i. e. without guide blades; let $v_1$ equal the entrance velocity AE of the water and $u_1$ be the velocity AF of the inner circle A.

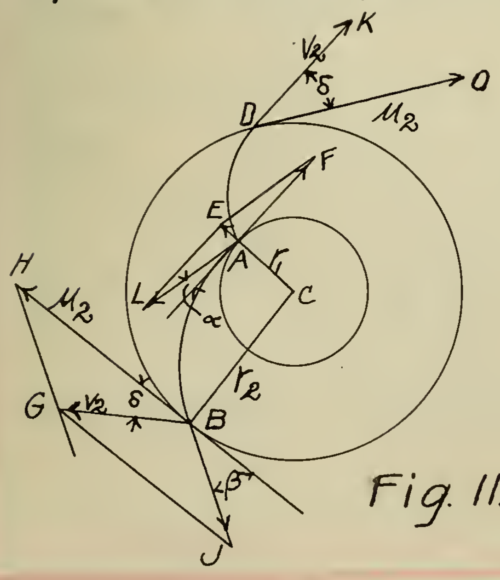


Fig. 11.

Now that a shock may be prevented, the first element of the blade at A must have a velocity in the direction AL equal to the initial velocity of the water relative to the rotating wheel. This condition

may be expressed by the equation $C^2 = V_1^2 + \mathcal{M}_1^2$. (1).
In consequence of the initial velocity $v_1$ of the water and the simultaneous rotation of the wheel, a particle of the water entering at A will traverse an absolute path represented, say by the curve AD, which cuts the outer circumference at the angle KDO $= \delta$. Let $v_2$ represent the absolute velocity with which the water leaves the wheel in the direction DK, and $u_2$ $\mathcal{M}_2$=DO = BH the tangential velocity of the wheel at the outer circumference; then the relative velocity $v_2$ with which the water moves along the last element B of the blade AB will be the resultant BJ of the absolute velocity GJ = HB= $-u_2$ equal and opposite to that of the wheel. The triangle GBJ therefore gives the equation

$$C^2 = V_2^2 + \mathcal{M}_2^2 - 2 V_2 \mathcal{M}_2 \cos. \delta. \qquad (2).$$

In addition to its velocity $v_1$ the water on entering the wheel at A has a certain hydraulic pressure $P_1$, which may be represented by a water column of the height $\frac{P_1}{\gamma}$ where $\gamma$ is the weight per unit volume of water, similarly $\frac{P_2}{\gamma}$ may represent the hydraulic pressure of the water leaving the wheel at B with the absolute velocity $v_2$. These pressures may be easily found. For, let b be the height of the water barometer, $\underline{h_1}$ the height of the suction estimated from the lower water-level up to the axis C, $\underline{h_2}$ the height of delivery measured from C to the upper water level $\underline{0}$ and let $S_1$ and $S_2$ be the heads corresponding to the resistances to motion in the suction and

delivery pipes respectively. As may be seen we then have

$$\frac{p_1}{\gamma} = b - h_1 - S_1 - \frac{V_1^2}{2g} \qquad (3).$$

Since the water that leaves the wheel with a velocity $V_2$ and a hydraulic pressure $p_2$ it must be capable of not only overcoming the delivery head $h_2$, the atmospheric pressure b and the resistance $S_2$, but also of maintaining the velocity of discharge w, we further have

$$\frac{p_2}{\gamma} + \frac{V_2^2}{2g} = h_2 + b + S_2 + \frac{w^2}{2g}. \qquad (4).$$

Now in order to determine the $u_2$ of the wheel which will give to the water the desired energy, it should be noted that owing to the centrifugal force the water in passing through the wheel from A to B has its living force increased by an amount corresponding to the head

$$\frac{u_2^2}{2g} - \frac{u_1^2}{2g}.$$

less the head $S_r$ which represents the resistance between the blades of the wheel. Accordingly, the action of the rotating wheel on the water passing through it, will be represented by the following equation:

$$\frac{p_2}{\gamma} + \frac{c_2^2}{2g} - \left(\frac{p_1}{\gamma} + \frac{c_1^2}{2g}\right) = \frac{u_2^2 - u_1^2}{2g} - S_r \qquad (5).$$

If we here substitute the values $c_1^2$, $c_2^2$, $p_1$ and $p_2$ given by equations (1) to (4), we have

$$h_1 + h_2 + S_1 + S_2 + \frac{w^2}{2g} - \frac{2V_2 u_2 \cos.\delta}{g} = -S_r.$$

Let $h = h_1 + h_2$ = total lift, and $S = S_1 + S_2 + S_r$ = sum of heads corresponding to hydraulic resistances, we get :-

$$h+\mathcal{S}+\frac{w^2}{2g}=\frac{V_2 u_2 \cos\delta}{g}. \qquad (6).$$

From this relation we can determine the velocity $u_2$, of the outer circumference of the wheel by combining it with the equation

$$V_2 = u_2 \frac{\sin\beta}{\sin(\beta+\delta)}.$$

obtained from the triangle BGH, thus finding

$$u_2 = \sqrt{g\left(h+\mathcal{S}+\frac{w^2}{2g}\right)\frac{\sin(\beta+\delta)}{\sin\beta\cos\delta}} = \sqrt{g\left(h+\mathcal{S}+\frac{w^2}{2g}\right)(1+\tan\delta\cot\beta)}. \qquad (7)$$

From this follows the absolute velocity with which the water flows from the wheel

$$V_2 = \sqrt{g\left(h+\mathcal{S}+\frac{w^2}{2g}\right)\frac{\sin\beta}{\cos\delta\sin(\beta+\delta)}} \qquad (8).$$

Equation (7) will enable us to compute the velocity of the outer circumference of the wheel, when $\delta, \beta, \mathcal{S}$, and w are known; according to Grove we may assume for $\delta$ and $\beta$

$$\tan\delta = .5; \qquad \delta = 26° 34'.$$

$$\tan\beta = .3; \qquad \beta = 16° 42'.$$

It is sufficiently accurate to place the head $\frac{w^2}{2g}$ due to velocity of discharge w equal to 3 per cent of the lift h. Further, if we assume in accordance with existing experimental results, the head S representing the resistances equal to .42h, we shall get for the velocity of the outer part of the wheel $u_2 = 1.4\sqrt{2gh}$, or about 40 per cent greater than the velocity which would be acquired in falling through the height h of the lift. From this follows the absolute velocity with which the water leaves the wheel

$$V_2 = .6\sqrt{2gh}.$$

If we denote the inner radius of the wheel by $r_1$ and the outer one by $r_2$ the number of revolutions per minute is expressed by

$$u = \frac{30u_2}{3.14r_2} = 9.55\frac{u_2}{r_2}.$$

The clear width of the wheel at the outer circumference can be found as follows: Let z represent the number of blades, s their normal thickness, and Q the amount of water that is to be discharged per second, then we have

$$Q = (2\pi r_2 \sin\beta - zs)b_2V_2.$$

In like manner for the entrance of the water we have the equation

$$Q = (2\pi r_1 - \frac{zs}{\sin\alpha})b_1V_1.$$

from which we may obtain either of the three quantities $b_1$, $v_1$, and $r_1$ when suitable values are assumed for the others.

We may further make the assumption that the area of the openings through which the water is admitted to the wheel is equal to the sectional area of the supply pipe, and hence, if we neglect the thickness of the blades, we have

$$\pi r_1^2 = 2\pi r_1 b_1$$

which gives for the clear width at the inner circle of the wheel

$$b_1 = \frac{r_1}{2}, \text{ etc.}$$

,After thus computing the angles $\alpha$, and $\beta$. which the blades make with inner and outer circumferences, it is necessary

to make some definite assumption from which to determine the form of the blades. This is due to the fact that, since the resistances experienced by the water in the wheel cannot be represented by analytical expressions, it is impossible to ascertain by calculation the most advantageous form of blade, or that which reduces the internal resistances to a minimum. If no such resistances were encountered, any form of blade intersecting the circumferences at the required angles $\alpha$ and B would evidently answer the purpose.

As a basis on which to proceed in designing the blades, suppositions have been made with respect to the law according to which the velocity of the water should vary in the wheel.

It has been supposed that the radial component of the velocity of the water should remain constant, and that the tangential velocity should increase according to a certain law and hence the absolute path of the water and finally the form of the blade has been determined, as in the turbines. The arbitrariness of all suppositions may justify us in assuming directly as simple a form as possible for the blades, say a circular arc that

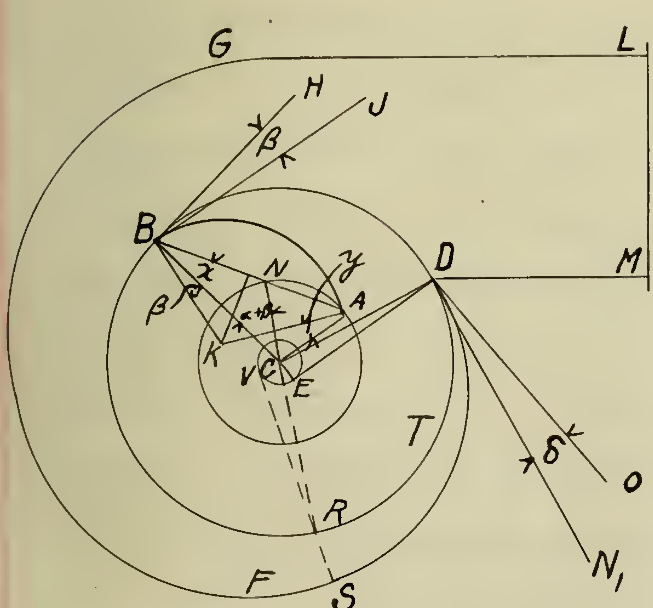



Fig. 11a.

cuts both circumferences at the desired angles $\alpha$ and $\beta$. To draw such an arc, lay off at any point B of the outer circumference a line BJ making with the tangent BH the angle HBJ = B, and at the point C lay off a line CN making with the radius CB an angle $BCN = \alpha + B$. This second line cuts the inner circumference at N, and if we connect this intersection with the point B and prolong the line thus obtained until it intersects the inner circle at a second point A, then the circular arc passed through A and B with a center K lying on the perpendicular to BJ at B will satisfy the required condition, as may be seen from the figure; for, the isosceles triangle CNA gives CNA = CAN or $\alpha + B + X = Y + B + X$, from which we obtain $CAK = Y = \alpha$.

The guiding apparatus which receives the water that leaves the circumference of the wheel at the angle $\delta$ and gradually leads it in the direction of the delivery pipe, is formed, in horizontal centrifugal pumps, by the casing, whose circumference may be regarded as a single guide blade. This casing is brought almost in contact with the wheel at a point D, and is here given the direction of the issuing water by making the angle NDO equal to $\delta$. A suitable form is obtained for the casing by shaping it as an involute DFG of a circle, which is concentric to the wheel and has the radius $CE = r_2 \sin \delta$. At any point R of the circumference of the wheel the sectional area RS furnished by the casing for the issuing water will be equal to the area of efflux DTR, which discharges the delivered water through RS. For if we denote the angle

$$DCR = ECV \quad \text{by } \phi$$

then

$$RS = \text{arc} \quad VE = r_2 \sin \delta \cdot \phi.$$

hence the sectional area offered to the water at this place is $b_2 r_2 \phi \sin \delta$, where $b_2$ is the clear width of the wheel. But this is equal to the area offered by the arc $DTR = r_2 \phi$, to the water issuing at the angle $\delta$, the diminution due to the area of the blades being neglected. The casing is provided with a neck GLMD which connects with the delivery pipe.

The power needed to drive the pump is equal to that required to lift a quantity of water Q per second to the height $h + S + \frac{w^2}{2g}$;

expressed in horse-power, this is given by

$$HP = \frac{Q\gamma}{75}\left(h + S + \frac{w^2}{2g}\right) = 13.33\,Q\left(h + S + \frac{w^2}{2g}\right) = .1149\,Q\left(h + S + \frac{w^2}{2g}\right).$$

As the water is actually lifted to the height <u>h</u> only, the efficiency of the pump will be expressed by

$$\eta = \frac{h}{h + S + \frac{w^2}{2g}}.$$

This value for efficiency does not include the journal friction.

The efficiency of a centrifugal pump, as stated before depends upon the care with which it is designed, and upon the conditions under which it is to operate. The first class is described above, viz: that type which is designed to work under special conditions and under those conditions only, is the most efficient. Designers seem to be able to attain to the highest efficiencies with pumps designed to operate under high heads; for, low head sewerage pumps have been designed

which give efficiencies of 65 per cent, and mine pumps under a head of 470 feet which give efficiencies of 76 per cent. The latter were built by Sulzer Brothers, Winterthur, Switzerland, for the silver mines at Horcajo, Spain. (1) The depth of the mine is 1280 feet. The pumping is done by compounded centrifugal pumps, driven by induction motors direct connected.

The pumps were installed at respective depths of 470 feet, 940 feet and 1280 feet below the mouth of the mine. The fourth unit being placed at the lowest level that is being worked. Each pumping unit is a four stage pump having four pressure wheels mounted on the same shaft and within the same case. The total pressure against which each unit works, running at 900 r. p. m., is about 240 pounds per square inch.

The construction is shown in Fig. 12 which at the left gives a longitudinal section through the center line of the

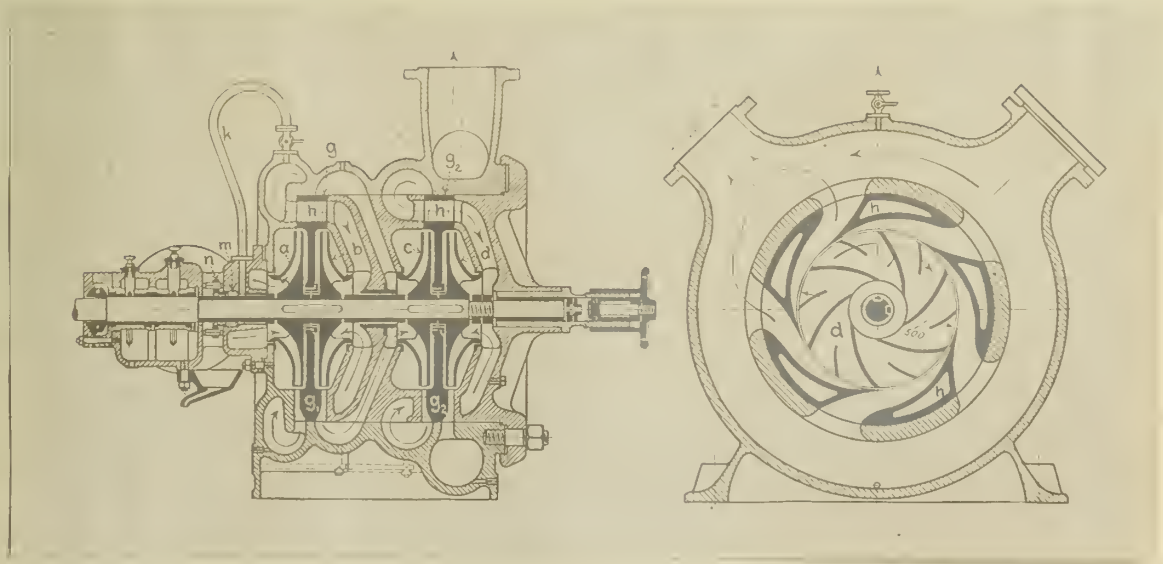



Fig 12.

(1). Engineering News, January 29, 1902.

shaft, and at the right, a transverse section through center line of the discharge casing. The four blade wheels marked a, b, c and d in the longitudinal section, have water passages curved as shown in the sections. The arrangement of the channels which lead the water from one wheel to another is rather intricate. Just outside each pair of wheels a, b and c, d is a casting $g_1g_2$, rigidly fastened to pump casing, in which spirally curved guide passages are formed, and as indicated by the arrows, the water flows outward through the guides into annular collecting chambers surrounding the respective wheels. From the collecting chambers separate sets of passages lead the water to the inner openings of the succeeding wheel. Between wheels b & c these passages are comparatively direct, but between wheels a & b and c & d the water must flow through channels h of triangular section, cored out in the guide castings $g_1$ and $g_2$ respectively, between the guide channels for the discharge from the wheel. The annular chamber receiving the discharge from the wheel d is larger than the others and is fitted with two outlet openings located on opposite sides, at angles of 45 degrees with the horizontal. In any particular place the more conveniently placed of these two outlets may be used, the other being closed by a cover plate. Similarly the suction end of the pump is formed as a tee, with two opposite openings so that no matter which side of the pump faces the piping no trouble will be experienced in making the connections.

The action of the pump exerts a resultant thrust toward the left end of the machine and to take this thrust, a ball bearing is provided, shown at the extreme left end of the shaft. To the right of this is a ring oiling journal bearing; the other bearing is formed in the cover of the pump on the right and is lubricated by a compression grease cup. Where the shaft passes through the casing at the left, it is surrounded by a tight bronze sleeve m, which passes through a stuffing box n. In the lowest pump of the series is a pipe k, connecting the first discharge chamber with the stuffing box, thus forming a liquid seal.

The maximum demand so far made upon the system has been 22 gallons per second, and the mean daily pumpage has been 1,425,000 gallons per 24 hours. The discharge below 15 gallons per second is controlled by throttling, and the discharge above that limit is regulated from the steam end. The tests, made on the system before the fourth pump was connected, showed an efficiency of 75.9 per cent. The entire installation has proved entirely satisfactory and the method a very efficient one for removing water from a mine.

This type of pump has been very little used in this country; this is probably due to the fact that the possibilities of the centrifugal pump have not been thoroughly appreciated by American engineers.

The Harris pump is another form of centrifugal pump for high heads. A description and discussion of this pump will

be given later in connection with the tests made upon it.

Perhaps the best example of a low head, high capacity pump, is of that type designed by Mr. C. P. Richards and Mr. Charles Bume, a Swiss engineer, for the drainage works of New Orleans. The conditions under which these pumps were to operate were :- A discharge of 37,000 gallons per minute at 125 r. p. m. under a three foot head. The high speed was required as the pumps were to be driven electrically. Owing to the high speed and the low head difficulties were immediately encountered when applying the ordinary type of centrifugal pump. The chief of these was the fact that the suction pipe was larger than the diameter of the runner. To obviate this difficulty a double suction pipe was provided and a double conoidal impeller took the place of the ordinary runner. By this means the discharge through the runner was decreased, and as a result the impeller diameter could be increased. The idea can be readily grasped by referring to the following diagram.

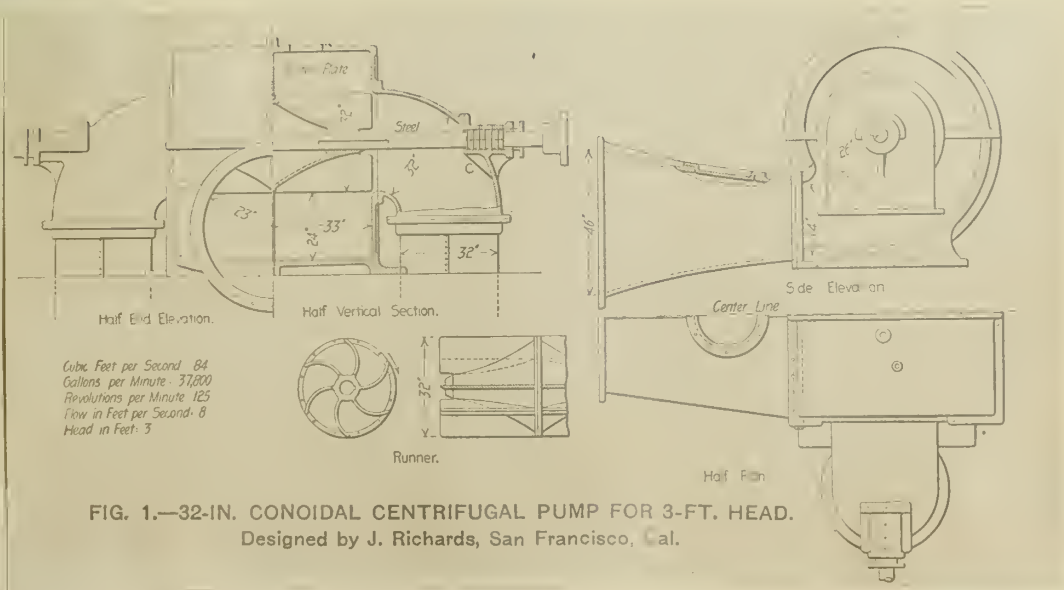



FIG. 1.—32-IN. CONOIDAL CENTRIFUGAL PUMP FOR 3-FT. HEAD.
Designed by J. Richards, San Francisco, Cal.

These pumps were very successful and when electrically driven at 125 revolutions per minute, and under three feet of head, the discharge was 37,800 gallons per minute, and the efficiency was found to be nearly 66 per cent.

## IV.- Tests Made by Writer

### a. - Centrifugal Pumps

The tests were made on four Gould centrifugal pumps, varying from 1-1/2 to 6 inches diameter of discharge. The objects of the test was to determine the efficiency of the pumps at various speeds and lifts.

The apparatus used in these tests was:- one suction gauge one discharge gauge, and one Hook gauge for measuring the height of the water above the crest of the weir. The weir used was 36 inches long with suppressed end contractions.

First a series of tests was made upon the engine,a 8"x10" Ball to determine the engine friction. From these results a curve was plotted from which the engine friction could be readily obtained for any given speed. Thus, the net horse power delivered to the pump was equal to the total indicated horse power, found from the indicator cards, minus the engine friction horse power for that speed. The engine was belted direct to the pump, which was placed over the large sump. The discharge passed through several valves in order that it could be throttled a little at each. This proceedure was necessary as the water passing through an opening at high pressure is apt to be affected by contraction of the stream,

hence a variable discharge would result. The discharge finally emptied into the weir channel,passed over the weir where its volume was measured,and then back into the sump

*36" Weir and Channel.*

The pump was primed by an ordinary steam jet ejector. The speed of the pump was given at all times by a belted tachometer. The engine revolutions were obtained with an ordinary revolution counter.

The pump, a Gould 6", has a 7" suction and a 6" discharge. The shape of the runner is best shown in Fig. 13.

Fig. 13.

The pressure gauge is used in the experiments where all calibrated by means of a Crosby testing machine, and curves were plotted from the result. The tachometer was calibrated by driving it from an electric motor and actually counting the revollutions of the motor by means of a counter.

The following data were taken; the readings on the suction and discharge gauges; the tachometer reading; the Hook gauge reading; the revolutions per minute of engine and indicator cards from the engine.

The total lift was equal to the actual distance between the surface of the water in the sump to the level of the flow from the discharge pipe, had the discharge been carried up in the air, plus the head due to the frictional resistances. Thus the total lift equals the sum of the readings on the suction and discharge gauges plus the distance between the two gauges.

The first series of tests was made at a constant total head of about fifteen feet and the speed was varied from about 350 to 550 r. p. m. In the second series the head was increased about 10 feet and the variations were made in speed. This plan was followed out until the total head reached about 60 feet. Three indicator cards were taken on most of the tests.

The hydraulic horse power is represented by the following equation:-

$$H.P. = \frac{D \times w \times h}{33000}.$$

where D equals discharge in gallons per minute; w equals the weight of one gallon of water; h equals total head in feet.

The horse power delivered to the pump is equal to the total horse power indicated by the engine minus the friction horse power of the engine for the given speed.

The efficiency of the pump is given by the following formula:

$$\eta = \frac{Net\ HP}{H\ HP}.$$

Test of

# Gould Pump No. 6.

## Summary of Results.

| Test No. | R.P.M. of Pump | Lift in feet. Suction | Disch'g | Total | Disch'g gals. per min. | Hyd. H.P. | Net H.P. | Eff. of Pump η. |
|---|---|---|---|---|---|---|---|---|
| 7 | 325 | 5.3 | 14.7 | 20 | 510 | 2.6 | 7.7 | 33.6 |
| 8 | 350 | 5.3 | 14.7 | 20 | 840 | 4.25 | 10.35 | 41 |
| 9 | 375 | 5.5 | 14.5 | 20 | 1145 | 5.8 | 13.55 | 42.8 |
| 10 | 400 | 6 | 14 | 20 | 1460 | 7.4 | 18.3 | 40.5 |
| 11 | 425 | 6 | 14 | 20 | 1640 | 8.3 | 21.35 | 39 |
| 12 | 450 | 5.3 | 14.7 | 20 | 1870 | 9.45 | 25.95 | 36.5 |
| 13 | 476 | 5.2 | 15.8 | 21 | 2010 | 10.7 | 30.1 | 35.5 |
| 14 | 500 | 5.2 | 17.3 | 22.5 | 2070 | 11.8 | 36.1 | 32.5 |
| 15 | 520 | 5 | 18 | 23 | 2250 | 13.1 | 40 | 32.7 |
| 16 | 475 | 5 | 25 | 30 | 1480 | 11.25 | 25.4 | 44.3 |
| 17 | 500 | 5 | 25 | 30 | 1710 | 13 | 31.1 | 41.8 |
| 18 | 450 | 6.5 | 24.5 | 30 | 1140 | 8.7 | 19.8 | 43.9 |
| 19 | 425 | 5 | 25 | 30 | 745 | 5.65 | 15.46 | 36.5 |
| 20 | 400 | 5 | 25 | 30 | 165 | 1.25 | 9.78 | 12.8 |
| 21 | 550 | 6 | 34 | 40 | 1605 | 16.15 | 37.9 | 42.5 |
| 22 | 540 | 6.3 | 33.7 | 40 | 1520 | 15.4 | 38.46 | 40 |
| 23 | 525 | 6.3 | 33.7 | 40 | 1315 | 13.3 | 32.57 | 40.5 |
| 24 | 500 | 6 | 34 | 40 | 940 | 9.5 | 26.22 | 36.2 |
| 25 | 475 | 5.5 | 34.5 | 40 | 345 | 3.5 | 17.7 | 19.8 |

# Test of
# Gould Pump No. 6.

## Summary of Results. (cont'd).

| Test No. | R.P.M. of Pump. | Lift in feet. | | | Disch'g. gals. per min. | Hyd. H.P. | Net H.P. | Eff. of Pump $\eta$. |
|---|---|---|---|---|---|---|---|---|
| | | Suction | Disch'g | Total. | | | | |
| 26 | 525 | 5.2 | 44.8 | 50 | 185 | 2.3 | 19.27 | 11.9 |
| 27 | 550 | 5.3 | 44.7 | 50 | 660 | 8.4 | 27.86 | 30 |
| 28 | 575 | 6 | 44 | 50 | 1290 | 16.4 | 39.4 | 41.6 |
| 29 | 600 | 6 | 54 | 60 | 610 | 9.25 | 35.2 | 26.3 |
| 30 | 400 | 5 | 11 | 16 | 1620 | 6.55 | 18 | 36.4 |
| 31 | 375 | 4.7 | 11.3 | 16 | 1325 | 5.35 | 13.4 | 40 |
| 32 | 350 | 5 | 11 | 16 | 1110 | 4.5 | 10.8 | 41.5 |
| 33 | 325 | 5 | 11 | 16 | 830 | 3.35 | 8.6 | 38.8 |
| 34 | 385 | 5.5 | 14.5 | 20 | 1275 | 6.45 | 15.4 | 42 |
| 35 | 400 | 5.8 | 14.2 | 20 | 1400 | 7.1 | 17.4 | 41 |
| 36 | 425 | 6 | 14 | 20 | 1670 | 8.45 | 20.9 | 40.5 |
| 37 | 440 | 6 | 24 | 30 | 1020 | 7.7 | 17.9 | 43 |
| 38 | 450 | 6 | 24 | 30 | 1125 | 8.5 | 18.1 | 47 |
| 39 | 462 | 6 | 24 | 30 | 1340 | 10.2 | 22.4 | 45.5 |
| 40 | 475 | 6 | 24 | 30 | 1460 | 11.1 | 24.3 | 45.5 |
| 41 | 500 | 6 | 24 | 30 | 1750 | 13.3 | 31.4 | 42.5 |
| 42 | 515 | 6 | 34 | 40 | 1150 | 11.7 | 26.8 | 43.5 |
| 43 | 525 | 6 | 34 | 40 | 1320 | 13.4 | 29.9 | 45 |
| 44 | 540 | 6 | 34 | 40 | 1530 | 15.5 | 35.1 | 43.5 |
| 45 | 550 | 6 | 34 | 40 | 1640 | 16.6 | 37.8 | 44 |

Tests Made on Small Gould Pumps.

The tests run on the smaller Gould pumps were carried out in the same manner as those on the 6" pump. Each pump when tested, was placed over a pit and driven by a crossed belt from the engine. This was necessary as there was no available room on the large sump, and when the small pump was placed over the pit as shown in figure 14 the belt had to be crossed in order to give the pump the proper direction of rotation. This fact probably caused a slight increase in the loss in transmission.

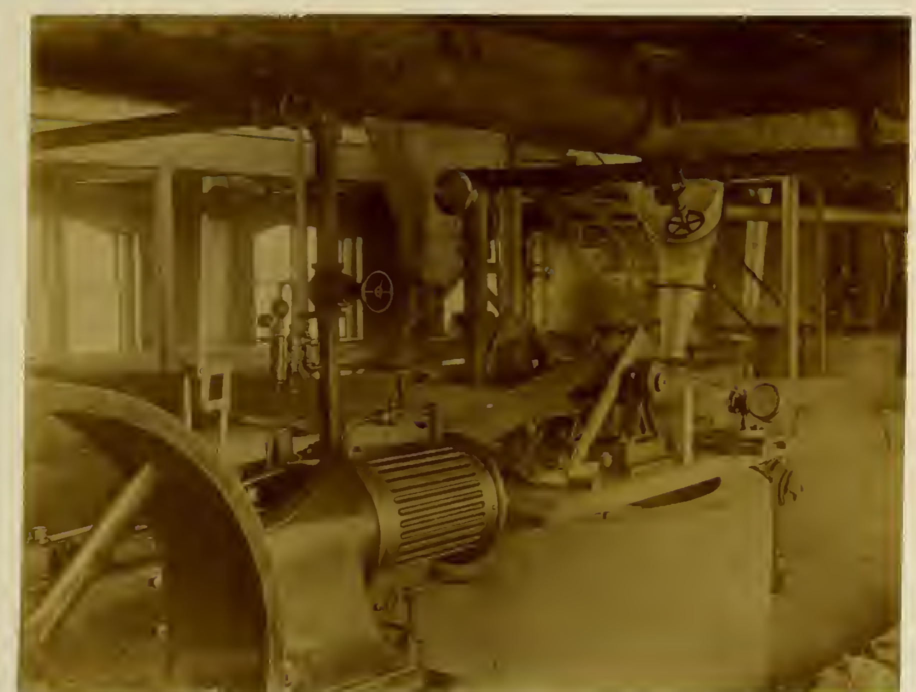


Fig. 14.

This method for measuring the power absorbed by the pump is less accurate for the transmission of small powers than it is for heavy loads. This is due to the fact that a small error in the indicator cards will cause a large error in

the final result. The pump discharge was carried upstairs where it was run over a 12" weir and in this way quantity was determined. The results were worked up in a manner similar to those of the previous experiments.

# TEST OF
# $1\frac{1}{2}$-inch Gould Pump.
## Summary of Results.

| No. | Lift in feet. | | | Gals. per min. Pumped. | R.P.M. Pump. | Horse Power | | Eff. % |
|---|---|---|---|---|---|---|---|---|
| | Disch'g. | Suction. | Total. | | | Hyd. | Net. | |
| 1 | $74\frac{1}{2}$ | $5\frac{1}{2}$ | 80 | 113 | 1900 | 2.3 | 7.33 | 31.2 |
| 19b | $75\frac{1}{2}$ | " | 81 | 90 | 1875 | 1.84 | 6.67 | 27. |
| 19a | $74\frac{1}{2}$ | " | 80 | 89 | 1875 | 1.80 | | |
| 20b | $74\frac{1}{2}$ | " | 80 | 85 | 1850 | 1.72 | 6.46 | 26.2 |
| 20a | $74\frac{1}{2}$ | " | 80 | 82 | 1850 | 1.66 | | |
| 21b | $74\frac{1}{2}$ | " | 80 | 77 | 1825 | 1.57 | 5.66 | 26.7 |
| 21a | $74\frac{1}{2}$ | " | 80 | 72 | 1825 | 1.45 | | |
| 5 | $64\frac{1}{2}$ | " | 70 | 142 | 1900 | 2.51 | 8.20 | 30.6 |
| 6 | $64\frac{1}{2}$ | " | 70 | 56 | 1700 | .99 | 5.17 | 19.2 |
| 4 | $56\frac{1}{2}$ | " | 62 | 157 | 1900 | 2.46 | 8.60 | 28.6 |
| 16a | $54\frac{1}{2}$ | " | 60 | 104 | 1700 | 1.59 | 5.73 | 27.8 |
| 16b | $54\frac{1}{2}$ | " | 60 | 104 | 1700 | 1.59 | | |
| 17a | $54\frac{1}{2}$ | " | 60 | 90 | 1650 | 1.37 | 5.12 | 26.8 |
| 17b | $54\frac{1}{2}$ | " | 60 | 90 | 1650 | 1.37 | | |
| 18a | $54\frac{1}{2}$ | " | 60 | 76 | 1600 | 1.15 | 4.30 | 26.8 |
| 18b | $54\frac{1}{2}$ | " | 60 | 75 | 1600 | 1.14 | | |
| 11 | $24\frac{1}{2}$ | " | 30 | 150 | 1560 | 1.14 | 6.56 | 17.4 |
| 10 | $24\frac{1}{2}$ | " | 30 | 142 | 1500 | 1.08 | 5.07 | 21.3 |
| 12 | $24\frac{1}{2}$ | " | 30 | 137 | 1500 | 1.04 | 5.25 | 19.8 |
| 9 | $24\frac{1}{2}$ | " | 30 | 123 | 1400 | .94 | 4.22 | 22.3 |
| 13b | $24\frac{1}{2}$ | " | 30 | 110 | 1350 | .84 | 3.65 | 22.8 |

# TEST OF
# $1\frac{1}{2}$ inch Gould Pump.

## Summary of Results (con't).

| No. | Lift in Feet | | | Gals. per min. Pumped | R.P.M. Pump | Horse Power | | Eff. % |
|---|---|---|---|---|---|---|---|---|
| | Disch'g. | Suction. | Total. | | | Hyd. | Net. | |
| 13a | $24\frac{1}{2}$ | $5\frac{1}{2}$ | 30 | 108 | 1350 | .82 | 3.65 | 22.8 |
| 14a | $24\frac{1}{2}$ | " | 30 | 99 | 1350 | .75 | 3.13 | 24.0 |
| 14b | $24\frac{1}{2}$ | " | 30 | 99 | 1300 | .75 | | |
| 15b | $24\frac{1}{2}$ | " | 30 | 87 | 1300 | .66 | 2.86 | 22.9 |
| 15a | $24\frac{1}{2}$ | " | 30 | 86 | 1250 | .65 | | |
| 8 | $24\frac{1}{2}$ | " | 30 | 95 | 1200 | .50 | 2.77 | 18.1 |
| 7 | $14\frac{1}{2}$ | " | 20 | 72 | 975 | .36 | 1.91 | 18.8 |

# TEST OF

# $1\frac{3}{4}$" GOULD PUMP.

## Summary of Results.

| No. | Lift in feet | | | R.P.M | Gals. P.M. | Horse Power | | Eff. |
|---|---|---|---|---|---|---|---|---|
| | Disch'g | Suction | Total | Pump | Pumped. | Hyd | Net. | % |
| 2 | 92 | 7 | 99 | 1625 | 200 | 5.02 | 15.6 | 32.2 |
| 15 | 73 | " | 80 | 1442 | 173 | 3.5 | | |
| 3 | 73 | " | 80 | 1425 | 167 | 3.4 | 9.9 | 34.2 |
| 14 | 73 | " | 80 | 1358 | 122 | 2.47 | 7.8 | 31.7 |
| 13 | 73 | " | 80 | 1302 | 50 | 1.00 | 5.6 | 18.1 |
| 12 | 73 | " | 80 | 1260 | No discharge. | | | |
| 11 | 52 | " | 59 | 1125 | 47 | .7 | 5. | 14 |
| 1 | 51 | " | 58 | | 222 | 3.25 | 15.7 | 20.7 |
| 8 | 55 | " | 62 | 1325 | 190 | 2.97 | 11.4 | 26.1 |
| 7 | 33 | " | 40 | 1138 | 180 | 1.82 | 6.3 | 28.9 |
| 4 | 35 | " | 42 | 1115 | 163 | 1.73 | 5.7 | 30.4 |
| 10 | 32 | " | 39 | 882 | 52 | .5 | 3.4 | 15 |
| 6 | 25 | " | 32 | 965 | 147 | 1.2 | 4.2 | 28.3 |
| 18 | 23 | " | 30 | 930 | 139 | 1.06 | 3.6 | 29.5 |
| 17 | 23 | " | 30 | 840 | 87 | .66 | 2.8 | 23.6 |
| 16 | 23 | " | 30 | 784 | 68 | .52 | 2.2 | 23.7 |
| 5 | 13 | " | 20 | 682 | 79 | .40 | 1.9 | 21.1 |
| 9 | 11 | " | 18 | 590 | 58 | .27 | 1.8 | 15 |

# TEST OF

# 2" GOULD PUMP.

## Summary of Results.

| No. | Lift in Feet | | | R.P.M Pump | Gals. P.M. Pumped. | Horse Power | | Eff. % |
|---|---|---|---|---|---|---|---|---|
| | Disch'g | Suction | Total | | | Hyd | Net. | |
| 10 | 76 | 4 | 80 | 1005 | 75 | 1.55 | 7.2 | 21.3 |
| 11 | 76 | " | 80 | 1020 | 120 | 2.45 | 8.9 | 27.5 |
| 12 | 76 | " | 80 | 1030 | 149 | 3.02 | 9.2 | 32.9 |
| 13 | 76 | " | 80 | 1050 | 150 | 3.04 | 9.3 | 32.7 |
| 14 | 76 | " | 80 | 1075 | 170 | 3.44 | 10.0 | 34.4 |
| 15 | 46 | " | 50 | 800 | 112 | 1.42 | 4.8 | 29.4 |
| 16 | 46 | " | 50 | 815 | 122 | 1.54 | 4.9 | 31.4 |
| 17 | 46 | " | 50 | 830 | 139 | 1.76 | 5.9 | 29.9 |
| 18 | 46 | " | 50 | 855 | 162 | 2.05 | 5.9 | 34.8 |
| 19 | 46 | " | 50 | 880 | 200 | 2.55 | 7.5 | 34 |
| 20 | 26 | " | 30 | 625 | 90 | .69 | 3. | 23 |
| 21 | 26 | " | 30 | 645 | 135 | 1.02 | 3.2 | 31.6 |
| 22 | 26 | " | 30 | 665 | 142 | 1.08 | 3.5 | 30.8 |
| 23 | 26 | " | 30 | | | | | |
| 28 | 33 | " | 37 | 675 | 97 | .91 | 3.3 | 27.6 |
| 29 | 81 | " | 85 | 1100 | 157 | 3.22 | 10.7 | 30.1 |
| 30 | 83 | " | 87 | 1050 | 71 | 1.56 | 7.1 | 22 |
| 31 | 59 | " | 63 | 1025 | 262 | 2.02 | 22.3 (?) | 9.1 (?) |
| 32 | 127 | " | 131 | 1287 | 43 | 1.43 | 6.0 | 23.9 |

[illegible]

# TEST OF

# 2½" GOULD PUMP.

## Summary of Results.

| No | Lift in Feet. | | | R.P.M | Gals. P.M | Horse Power | | Eff. |
|---|---|---|---|---|---|---|---|---|
| | Disch'g. | Suction | Total | Pump. | Pumped | Hyd. | Net. | % |
| 28 | 98 | 4 | 102 | 1020 | 102 | 2.64 | 16 | 16.5 |
| 27 | 81 | " | 85 | 960 | 186 | 4.00 | 14.2 | 28.2 |
| 4 | 66 | " | 70 | 935 | 313 | 5.60 | 16.2 | 34.6 |
| 26 | 66 | " | 70 | 920 | 291 | 5.19 | 15.2 | 34.4 |
| 25 | 66 | " | 70 | 900 | 245 | 4.26 | 12.5 | 34.1 |
| 3 | 66 | " | 70 | 890 | 220 | 3.95 | 12.1 | 32.6 |
| 24 | 66 | " | 70 | 880 | 162 | 2.9 | 9.9 | 29.3 |
| 23 | 66 | " | 70 | 865 | 140 | 2.5 | 9.8 | 25.5 |
| 2 | 66 | " | 70 | 860 | 120 | 2.15 | 9.4 | 22.9 |
| 22 | 66 | " | 70 | 850 | — | — | — | — |
| 5 | 46 | " | 50 | 875 | 340 | 4.33 | 17. | 25.5 |
| 21 | 46 | " | 50 | 835 | 330 | 4.18 | 13.1 | 31.9 |
| 20 | 46 | " | 50 | 820 | 304 | 3.9 | 10.5 | 37.1 |
| 6 | 46 | " | 50 | 790 | 237 | 3.0 | 9.3 | 32.3 |
| 19 | 46 | " | 50 | 770 | 216 | 2.75 | 7.6 | 36.2 |
| 18 | 46 | " | 50 | 760 | 207 | 2.65 | 7.0 | 37.9 |
| 17 | 46 | " | 50 | 750 | 137 | 1.7 | 6.1 | 27.9 |
| 7 | 46 | " | 50 | 740 | 120 | 1.55 | 6.5 | 23.9 |
| 16 | 46 | " | 50 | 735 | 82 | 1.06 | 6.2 | 17.2 |

# TEST OF

# $2\frac{1}{2}$" GOULD PUMP.

## Summary of Results. (Cont.)

| No. | Lift in feet. | | | R.P.M. Pump | Gals. p.m. Pumped. | Horse Power | | Eff. % |
|---|---|---|---|---|---|---|---|---|
| | Disch'g | Suction | Total | | | Hyd. | Net. | |
| 15 | 46 | 4 | 50 | 725 | 30 | .38 | 5.1 | 7.45 |
| 12 | 26 | 4 | 30 | 640 | 242 | 1.85 | 5.9 | 31.4 |
| 13 | 26 | 4 | 30 | 610 | 202 | 1.55 | 5.8 | 26.7 |
| 11 | 26 | 4 | 30 | 595 | 186 | 1.43 | 4.3 | 33.3 |
| 10 | 26 | 4 | 30 | 580 | 146 | 1.12 | 3.8 | 29.5 |
| 9 | 26 | 4 | 30 | 565 | 99 | .77 | 3.2 | 23.4 |
| 8 | 26 | 4 | 30 | 550 | 96 | .73 | 3.1 | 23.6 |
| 35 | 16 | $2\frac{1}{4}$ | $18\frac{1}{4}$ | 946 | 551 | 2.56 | 19.7 | 13. |
| 34 | $16\frac{1}{4}$ | $2\frac{1}{4}$ | $18\frac{1}{2}$ | 950 | 551 | — | — | — |
| 32 | $13\frac{3}{4}$ | $3\frac{1}{2}$ | $16\frac{3}{4}$ | 897 | 460 | 1.96 | 15. | 13.05 |
| 33 | 11 | $2\frac{1}{4}$ | $13\frac{1}{4}$ | 780 | 440 | 1.48 | 10.9 | 13.5 |
| 29 | $9\frac{1}{4}$ | $2\frac{1}{4}$ | $11\frac{1}{2}$ | 700 | 394 | 1.14 | 8.4 | 13.6 |
| 31 | $7\frac{1}{2}$ | $2\frac{1}{4}$ | $9\frac{3}{4}$ | 634 | 353 | .87 | 6.3 | 13.8 |
| 30 | $6\frac{1}{4}$ | $2\frac{1}{4}$ | $8\frac{1}{2}$ | 551 | 300 | .65 | 4.9 | 13.3 |

Tests Made on Harris Air Bound Centrifugal Pump.

The Harris pump, Fig. 15 is of the vertical type, having an 8" suction and discharge pipe. The runner is 14" in diameter and its construction can be understood from Fig. 16.


S.

Fig. 15.


Fig. 16.

At a is represented the runner, a hemispherical shaped casting with its convex side up; at c are the vanes which catch the water and throw it from the suction s up into the hollow of the runner which is divided into compartments by radial partitions. These partitions give the water the same velocity as the runner. The velocity produces centrifugal force acting outward and as the runner is concave the water is driven out and down through the passage b, which lies just over the passage d. In this passage are interposed vanes e, which gradually slant downward and change the resultant angular velocity, caused by the runner, into pressure head which forces the water out the discharge. Above the runner a is a chamber in which air is pocketed.

The inventor undertook in the design of this pump to raise the efficiency of the centrifugal pump by pocketing air above the runner thus lowering the friction loss. This should result as the loss from friction would be less when the water rubs upon air than when water rubs upon iron.

The pump was placed over the large sump and belted to the engine by a quarter twist belt. The suction pipe was supplied with a foot valve which was used to vary the suction lift. The discharge was carried to the weir channel and measured as it passed over the 36" weir. A small valve was placed on a pipe connected to the suction pipe. Air was sniffed in through this valve, carried through the pump, trapped out of the discharge into a chamber and forced back into the pump above the runner. A vacuum gauge was placed on the suction, a

pressure gauge on the discharge, and another pressure gauge was connected to the chamber over the runner. The majority of the tests was made at 750 or 900 r. p. m.

The desired head was obtained, the speed was kept constant until the flow became uniform; then readings were taken from the suction, discharge, air chamber, and Hook gauges, and indicator cards and revolutions per minute were taken from the engine.

The results were worked up as in the previous tests.

# TEST OF

## Harris Air Bound Pump.

### Summary of Results.

| No. | R.P.M. Pump | Lift in feet | | | | Gals. p.m. Pump'd | Horse Power | | Eff. % |
|---|---|---|---|---|---|---|---|---|---|
| | | Chamber | Suction | Disch'g | Total. | | Hyd | Net. | |
| 71 | 900 | $25\frac{1}{2}$ | 11.1 | $16\frac{1}{2}$ | 27.6 | 1140 | 7.94 | 24.7 | 32.1 |
| 65 | " | $21\frac{1}{2}$ | 9 | 18 | 27 | 1085 | 7.42 | 24.45 | $30\frac{1}{2}$ |
| 15 | " | 21 | 11.2 | 15 | 26.2 | 1075 | 7.4 | 23.4 | 31.6 |
| 72 | " | $28\frac{1}{2}$ | 9.9 | $25\frac{1}{2}$ | 35.4 | 930 | 8.08 | 22.7 | 35.5 |
| 66 | " | 27 | 9 | $25\frac{1}{2}$ | 34.5 | 930 | 8.1 | 22 | 36.8 |
| 36 | " | 22 | 11.8 | $17\frac{1}{2}$ | 29.3 | 895 | 6.6 | 17.3 | 38.2 |
| 16 | " | 24 | 10.1 | $21\frac{1}{2}$ | 31.6 | 858 | 6.8 | 21.4 | 31.8 |
| 73 | " | 29 | 7.8 | $30\frac{1}{2}$ | 38.3 | 790 | 7.64 | 21.5 | 35.5 |
| 67 | " | 28 | 7.9 | $30\frac{1}{2}$ | 38.4 | 790 | 7.66 | 20.9 | 36.6 |
| 74 | " | 30 | 7.2 | $35\frac{1}{2}$ | 42.7 | 685 | 6.87 | 19.9 | 34.5 |
| 68 | " | 30 | 7.7 | $35\frac{1}{2}$ | 43.2 | 685 | 7.47 | 20.9 | 35.7 |
| 38 | " | $27\frac{1}{2}$ | 10.1 | 35 | 45.1 | 610 | 7.1 | 15 | 47.4 |
| 75 | " | 31 | 6.6 | $42\frac{1}{2}$ | 49.1 | 515 | 6.24 | 18.6 | 33.5 |
| 69 | " | 33 | 6.6 | $41\frac{1}{2}$ | 48.1 | 400 | 4.86 | 20.1 | 24.2 |
| 39 | " | 28 | 8.4 | $36\frac{1}{2}$ | 44.9 | 350 | 4 | 10.1 | 39.2 |
| 35 | 880 | 15 | 10 | 1[illegible] | 25 | 960 | 6 | 17.5 | 34.3 |
| 37 | 890 | $24\frac{1}{2}$ | 10.6 | 29 | 39.6 | 692 | 6.9 | 15.2 | 45.5 |
| 40 | 890 | — | 8.4 | $34\frac{1}{2}$ | 42.9 | 325 | 3.5 | 8 | 43.8 |
| 41 | 887 | 12 | 10.1 | 27 | 37.1 | 610 | 5.7 | 13.5 | 42.3 |
| 21 | 760 | $13\frac{1}{2}$ | $7\frac{1}{4}$ | 24 | 31.3 | 152 | 1.2 | 7.6 | 15.8 |

# TEST OF

# Harris Air Bound Pump.

## Summary of Results (Cont.)

| No. | R.P.M. Pump | Lift in feet. Chamber | Suction | Disch'g | Total | Gals. p.m. Pump'd | Horse Power Hyd. | Net. | Eff. % |
|---|---|---|---|---|---|---|---|---|---|
| 22 | 760 | $13\frac{1}{2}$ | $7\frac{1}{4}$ | $24\frac{1}{2}$ | $31\frac{3}{4}$ | 145 | 1.2 | 7.6 | 15.8 |
| 23 | 760 | 5 | $7\frac{1}{4}$ | $17\frac{1}{2}$ | $24\frac{3}{4}$ | 130 | .8 | 7.7 | 10.4 |
| 54 | 750 | 15 | 10.1 | 9.5 | 19.6 | 910 | 4.5 | 14.7 | 30.6 |
| 55 | " | 16.5 | 10.1 | 9.5 | 19.6 | 780 | 4.9 | 12.6 | 38.9 |
| 60 | " | 16 | 8.9 | 10 | 18.9 | 900 | 4.45 | 14.4 | 31 |
| 8 | " | 15 | 9.2 | 10.5 | 19.7 | 775 | 3.9 | 13.1 | 29.7 |
| 9 | " | 14 | 8.9 | 10.5 | 19.4 | 770 | 3.8 | 13.1 | 29 |
| 17 | " | 13 | 9.5 | 10.5 | 20 | 738 | 3.7 | 12.5 | 29.5 |
| 61 | " | 16 | 7.7 | 15.5 | 23.2 | 685 | 4 | 12.1 | 33 |
| 10 | " | 15 | 8.4 | 15 | 23.4 | 675 | 4 | 12.2 | 32.7 |
| 27 | " | 14 | 9.5 | 15.5 | 25 | 640 | 4 | 12.1 | 33 |
| 28 | " | 13 | 9.2 | 15 | 24.2 | 630 | 3.9 | 11.3 | 34.5 |
| 25 | " | 12 | 10.1 | 14.5 | 24.6 | 630 | 3.9 | 12 | 32.5 |
| 24 | " | 11.5 | 10.4 | 14.5 | 24.9 | 630 | 4 | 12 | 33.3 |
| 56 | " | 18 | 7.8 | 19.5 | 27.3 | 625 | 4.3 | 12.2 | 35.2 |
| 57 | " | 18 | 7.2 | 22.5 | 29.7 | 580 | 4.3 | 12.5 | 34 |
| 26 | " | – | 1.5 | 12 | 13.5 | 570 | 3.1 | 12.2 | 35.4 |
| 11 | " | 15 | 7.8 | 17.5 | 25.3 | 560 | 3.6 | 11.3 | 31.8 |
| 18 | " | 14 | 8.7 | 17.5 | 26.2 | 560 | 3.7 | 11.4 | 32.5 |
| 62 | " | $16\frac{1}{2}$ | 6.9 | 20 | 26.9 | 560 | 3.8 | 11.6 | 32.8 |

# TEST OF
## Harris Air Bound Pump.
### Summary of Results. (Cont.)

| No. | R.P.M. Pump. | Lift in feet. Chamb. | Suction | Disch'g | Total | Gals p.m. Pump'd | Horse Power. Hyd. | Net. | Eff %. |
|---|---|---|---|---|---|---|---|---|---|
| 63 | 750 | 14½ | 5 | 25 | 30 | 505 | 3.35 | 10.87 | 38 |
| 12 | 750 | 15½ | 7.5 | 20½ | 28 | 490 | 3.4 | 10.65 | 32 |
| 13 | 750 | 16 | 7¼ | 22½ | 29¾ | 430 | 3.2 | 10.53 | 30.5 |
| 14 | 750 | 17½ | 6.7 | 24½ | 31¼ | 360 | 3 | 10.22 | 29.4 |
| 19 | 750 | 15 | 7.8 | 20½ | 28.3 | 415 | 2.97 | 9.94 | 33.5 |
| 20 | 880 | 16 | 7¼ | 22½ | 29¾ | 380 | 2.85 | 9.3 | 32.5 |
| 64 | 750 | 21 | 4¼ | 35 | 39¼ | 160 | 1.6 | 9.07 | 17 |

b. Tests Made on Rotary Pump.

The pump tested under this head was a 10" Johnson rotary Pump. This pump is a development of the old Pappenheim principle, that of two gears meshing together inside a casing. In this case, one gear-a large one - carries four teeth placed at intervals of 90° around its periphery. The other gear - a small one - has a cavity into which one of the teeth will fit. This gear turns four times as fast as the large one, hence always bringing the cavity into mesh with the tooth. The principle can readily be understood from Fig. 17.

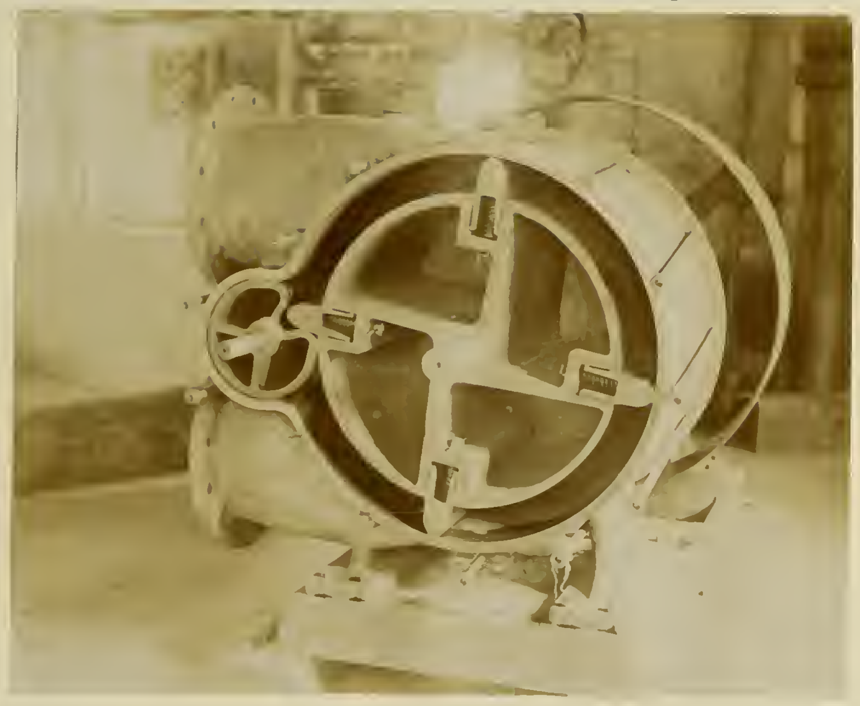


Fig. 17.

The pump was placed over the large sump and the discharge was passed into the three foot weir channel. There was no discharge head as the water flowed out of the discharge of the

pump against no hydrostatic head. Hence, the total head is equal to the suction head plus the velocity head. The latter item was found to be small as the suction and discharge pipe were large for the capacity of the pump.

The theoretical volume of discharge of the pump is equal to the effective area of the tooth times the mean diameter of the shell and the cylinder times $\pi$. Or the discharge in gallons per revolution is:

$$Q = \frac{23.81 \times 44.1 \times \pi}{231} = 14.27 \text{ gals.}$$

The actual discharge will be somewhat less than the theoretical as there will be some loss from slip.

The experiment was carried out in a manner similar to that followed in the previous tests.

# TEST OF

## Johnson Rotary Pump.

### Summary of Results.

| No. | R.P.M. Pump. | Total Lift | Gals. p.m. Pumped. | Horse Power. | | Eff. % |
|---|---|---|---|---|---|---|
| | | | | Hyd. | Net. | |
| 1 | 63 | 8 | 850 | 1.7 | 6.26 | 27.2 |
| 2 | 58 | 8½ | 810 | 1.73 | 5.46 | 31.6 |
| 3 | 96 | 10 | 1050 | 2.34 | 10.7 | 21.5 |
| 4 | 76 | 9½ | 1010 | 2.3 | 10.37 | 22 |
| 5 | 102 | 11 | 1290 | 2.53 | 12.59 | 20.2 |
| 6 | 100 | 11 | 1280 | 2.52 | 12.89 | 19.6 |
| 7 | 100 | 11 | 1260 | 2.51 | 11.4 | 22.1 |
| 8 | 108 | 11½ | 1305 | 2.54 | 13.63 | 18.6 |

## V. Conclusion

From the results of the tests made on the Gould Centrifugal pumps this conclusion can be drawn:-

The 6" pump, showing a maximum efficiency of 46%, is a good commercial pump; and its efficiency is about as high as could be expected for a pump of the general commercial type.

The smaller pumps tested, showing maximum efficiencies varying from 30 to 35%, compare very favorably with the pump of larger size. The comparison would have been still more favorable had some exact means for measuring the net horse power, delivered to the pump been at hand. In this case a considerable amount of power was lost in the belt transmission as the belt used was a heavy, crossed double belt.

The tests made on the Harris Pump showed it to be a slightly more efficient pump than the Gould. The effect of air over the runner was thoroughly investigated with the result that a slight increase in efficiency was shown. But the effect was not as marked as it was thought that it would be.

The tests made on the Johnson Rotary pump were not as extensive as at first planned. This was due to the fact that the pump was not tested before leaving the factory, and when it was started up in the laboratory, the impellers did not come

into exact mesh with the cavity on the small cylinder; hence, a hammer blow occurred as each tooth meshed. This blow at times became dangerous and the tests finally had to be discontinued. The efficiency obtained in the tests was very low for a rotary pump. This was natural, as there was necessarily much slip, for each time a tooth meshed the tooth was driven back into its slot in the cylinder; hence allowing the water to pass by the ends of the teeth back into the suction. The hammer blow also absorbed much energy which would otherwise have been utilized in driving the water.

Finally, after a careful consideration of the facts in regard to the two types of pumps (Centrifugal and Rotary) one is led to the conclusion that the centrifugal pump, although less efficient, is simpler, has fewer wearing parts, and is cheaper than the rotary pump; and that, considering these facts, the centrifugal type is the more economical in the long run, to purchase and operate.